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A New Algorithm for Optimal Design of the Recirculating Cooling Water System of Thermal Power Plants Part I: Description of the Methodology & Case Study 1

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Abstract

An innovative approach to the optimization of process parameters and equipment sizes of the recirculating cooling water system for various types of thermal power plants (TPPs) with natural draft wet cooling towers is presented in this paper. The optimal values of the most influential operating and dimensional parameters of a TPP cooling water system are obtained by minimizing the annual cost of the system while satisfying the specified input design and operating conditions as well as the imposed constraints. The specificities of the TPP type are determined through the input parameter for levelized cost of energy (LCOE), which also reflects the specificities of the environmental protection standards for each TPP type. The developed mathematical model and the computer program are cross-checked with various existing designs, and the results are found to be compliant and accurate. The proposed optimization method has a global character because the climatic and economic specificities of the geographic location of the TPP are determined through the input parameters. By applying the proposed optimization model, it is possible to make significant savings in the operation of a TPP on an annual basis. This article is organized into several parts to illustrate the application of the proposed optimization method using case studies.

Keywords

Thermal Power Plant, Cooling Water System, Cold End System, Natural Draft Cooling Tower, Steam Condenser, Optimization

1. Introduction

Plants that produce electricity via the conversion of heat energy (obtained in different ways) are generally referred to as thermal power plants (TPPs). This energy conversion is accomplished in a thermodynamic closed-loop process known as the Rankine cycle or the Clausius-Rankine (C-R) cycle. Figure 1. shows the main components of the C-R cycle.

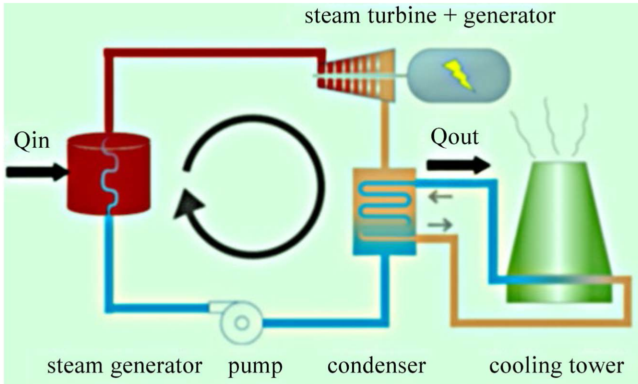

Figure 1. Components of the C-R cycle.

As shown in Figure 2, in the C-R cycle with superheated steam, the heat input is realized along parts of the cycle: heating the feedwater to the saturation temperature (line ab), evaporating the water at a constant temperature (line bc), and superheating the steam to higher temperatures (line cd).

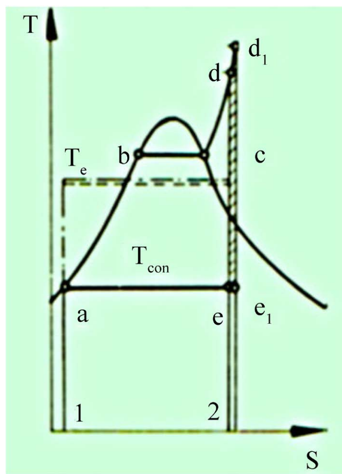

Figure 2. T-s diagram of the C-R cycle [1].

When considering the influence of certain basic thermodynamic parameters on the thermal efficiency of the C-R cycle, it is convenient to replace this cycle with the equivalent Carnot cycle.

The amount of heat supplied to the cycle is determined by the integral, taken in the area of entropy change from s_1 to s_2, which can also be represented as the product of an equivalent temperature (T_e) and the entropy difference ($s_2 - s_1$) [1].

$$q_{\text{in}} = \int_{s_1}^{s_2} T \cdot \text{d}s = T_\text{e} \cdot \left(s_2 - s_1\right) \tag{1}$$

The equivalent temperature is the mean temperature of the heat supply to the cycle, at which the thermal efficiency of the C-R cycle (η_{CR}) is equal to the thermal efficiency of the equivalent Carnot cycle (η_C), which allows us to write [1]:

$$\eta_{\text{C-R}} = \eta_\text{C} = \frac{T_\text{e} - T_{\text{cond}}}{T_\text{e}} \tag{2}$$

$$T_\text{e} = \frac{T_{\text{cond}}}{1 - \eta_{\text{C-R}}} \tag{3}$$

The thermal efficiency of the C-R cycle used in steam power plants typically falls within the range of 30% to 45%. There are several ways to improve the thermal efficiency of the C-R cycle: working with superheated steam, increasing the temperature and pressure of superheated steam, intermediate reheating of steam, reduction in steam condensing pressure, and regenerative heating of steam condensate [1].

The essence of all the improvements mentioned comes down to the general principle according to which increasing the thermal efficiency of the C-R cycle can be achieved by increasing the temperature of the heat source and/or lowering the temperature of the heat sink.

When it comes to the C-R cycle improvements related to increasing the temperature of the heat source, the process can be said to be complete for the current metallurgical capabilities of the materials. The parameters of live and reheated steam are optimized and internationally harmonized. The production of steam boilers and turbines is standardized on this basis.

Lowering the temperature of the heat sink in technical systems is limited by the ambient temperature. The ambient temperature is determined by the climate and geographical location of the region where the process takes place and cannot be influenced. For this reason, the standardization of a part of a TPP whose operation depends on the ambient temperature is not possible, because each plant construction location has its own climatic specificities that differ significantly from region to region.

The improvement of the thermal efficiency of the C-R cycle, with a decrease in the steam condensing pressure (p_{cond}), *i.e.*, with a decrease in the steam condensing temperature (T_{cond}), essentially came down to determining how close the steam condensation temperature can be to the ambient wet-bulb temperature ($T_{\text{wb-amb}}$).

As seen in **Figure 3**, to obtain a lower steam condensation temperature for a given $T_{\text{wb-amb}}$, the initial temperature difference (ΔT_{ITD}) and the approach (ΔT_{app}) can be reduced. This can be achieved by either increasing the performance of the cooling water system, increasing the heat transfer surface area in the condenser, increasing the size (diameter and shell height) of the cooling tower, increasing the fill volume in the cooling tower, installing more effective fill material, reducing the flow losses, improving the rain zone performance, and/or increasing the cooling water mass flow rate, thus reducing the cooling range (ΔT_{cw}) [2].

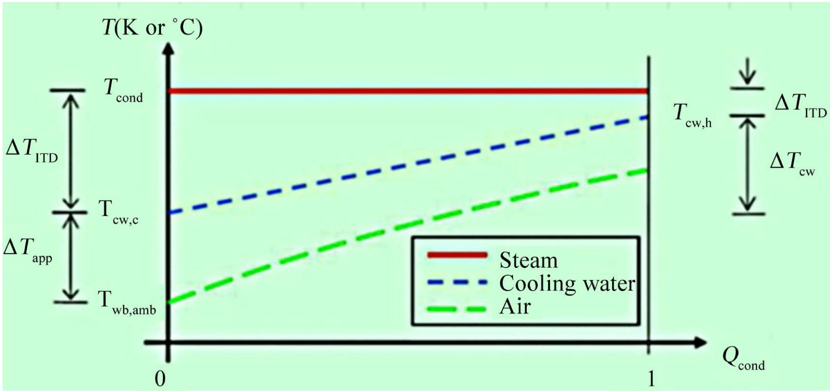

Figure 3. Schematic T-Q diagram for a wet-cooled power plant cooling water system [2].

Based on the above stated, it can be concluded that the TPP cold end system, including the low-pressure steam turbine (LPST), steam condenser (SC), cooling tower (CT), circulating water pumps (CWPs) and circulating water pipelines (CWPLs) remains the only part of TPPs whose parameters and dimensions are not subject to standardization.

The climatic and economic specificities of the geographical location of the TPP are the primary factors that should determine the characteristics of the cold end system in each project. In practice, this is not always the case. The construction of TPPs (including elements of the cold end system) is often based on the import of equipment and loans. The optimization of the TPP cold end system is either not done at all, or partial optimization is carried out that does not give satisfactory results. One of the main reasons for this is the lack of appropriate optimization models that include the entire TPP cold end system. This was the motivation to initiate a research project with the aim of developing a numerical multi-parameter optimization model for the TPP cold end system, applicable for various types of TPPs that can be used worldwide.

2. Literature Review

There are many research papers dealing with the theory, principles of operation, and performance analysis of steam turbines, steam condensers, cooling towers, and cooling water systems for various types of TPPs. However, there are a relatively small number of research papers dealing with their cost-optimal design and operation, most of which deal only with partial optimization that includes a small number of decision variables. To the best of the authors' knowledge, very few research papers have been published dealing with the complex multi-parameter optimization of the entire TPP cold end system, including the LPST, SC, CT and CWPs and CWPLs. There are several reasons for this current situation:

1) Carrying out a multi-parameter optimization of any engineering system is a

very complex task. Creating an appropriate mathematical model, economic model, and computer program requires expert knowledge in several different scientific fields: the field of engineering, the field of mathematics, and the field of economics. Each of these fields is individually very complex.

2) The application of analytical methods for multi-parameter optimization is a very complex task, even when the objective function is a function of a small number of decision variables. This becomes almost impossible when the objective function includes a greater number of decision variables. Introducing constraints and simplifications, which are often necessary to arrive at a solution, can compromise the accuracy of the optimization.

3) The application of numerical methods for carrying out a multi-parameter optimization, with the use of computers, until recently was limited by the capabilities of computers to find optimal solutions in a reasonable time.

4) Complex optimization of a cooling water system requires knowledge of the equipment cost functions; these should be a direct or indirect function of the decision variables and be based on reliable data. The best information for the equipment price functions is owned by equipment suppliers. However, this information is usually not available for public use. This is particularly characteristic of natural draft cooling towers, which have reached dimensions that make them one of the largest structures ever built with very large investments. The lack of ability to arrive at reliable equipment price functions can compromise the accuracy of the optimization.

Bearing in mind the above-stated reasons, it is understandable why the optimization of process parameters and equipment dimensions of a TPP cold end system was often focused on the system components only (CT, SC, or CWPs) and sometimes only on individual parameters that influence their operation. These include the effect of the SC inlet cooling water temperature [3], the effect of stem condensing pressure [4]-[6].

The research papers [7]-[12] are examples of partial optimization studies. The limitation of these types of studies is that the optimization results do not reflect the entire system and therefore have limited use value. Additional problems that often arise in partial optimizations are the interactions between sub-components of the system and the need to properly consider the influence of other parts of the system on the part under consideration.

Reference [13] studied the performance of the power plant with the combination of dry and wet cooling systems in different operating conditions. Then the off-design behavior was studied by varying the ambient temperature and relative humidity and several parameters connected to the cooling performance, like the exhaust steam flow rate, the air-cooling fan load and the number of operating cooling water pumps and cooling towers.

Instead of developing optimization algorithms that are specific for the TPP cold end components, some researchers preferred using general-purpose (open-source) algorithms such as the Genetic Algorithm (GA) [8], the Constrain Variable Metric Algorithm (CVMA) [9], and the Artificial Cooperative Search (ACS) algorithm [11]. These types of algorithms have advantages and disadvantages, some of which are listed in the referenced literature.

H. Kunaj and D. Barilar [14] presented a method for TPP cold end system optimization that was developed in the Institute of Energy Zagreb and used for NPPs Krško and Prevlaka. A. Popović [15] [16] presented a method for TPP cold end optimization that was developed in the Institute of Thermotechnics and Nuclear Engineering—ITEN, Energoinvest Sarajevo. The method provides optimal dimensions of dry and wet natural draft cooling towers for 100 MW to 500 MW TPPs. The objective function for the optimization methods presented in [14] and [15] [16] was the minimum price of the produced electric energy and applicable for economic specificities and the geographic location of the Yugoslavia state at the time of the studies. The limitation of these types of studies is that the optimization results do not have a global character and cannot be applied to all types of TPPs.

This study aims to fill the research gap and address the limitations of previous studies. The hypothesis that is put forward is the development of a mathematical model and a computer program specific to the cold end system for various types of TPPs that can be used worldwide.

3. Decision Variables

Among numerous operating and dimensional parameters of the TPPs cold end system components, the following seven decision variables were selected as the most suitable as optimal design control variables: cooling water approach to the ambient wet bulb temperature (ΔT_{app}), cooling water range (ΔT_{cw}), steam condenser terminal temperature difference (ΔT_{TTD}), cooling water velocity in the steam condenser tubes (v_{SCt}), hydraulic water load on the cooling tower fill (q_{CTf}), height of the cooling tower fill (H_{CTf}), and height of the cooling tower air inlet opening (H_{CTi}).

4. Objective Function

To optimize the decision variables of the cooling water system in TPPs, it is most suitable to take the annual cost of the cooling water system (AC_{CWS}) as the objective function and to optimize the process and dimensional parameters of the system so that the AC_{CWS} is minimal. The AC_{CWS} can be expressed as the sum of the annual investment cost of the cooling water system (AIC_{CWS}) and the annual operating cost of the cooling water system (AOC_{CWS}).

$$AC_{\text{CWS}} = AIC_{\text{CWS}} + AOC_{\text{CWS}} \tag{4}$$

The capital cost of the cooling water system (CC_{CWS}) can be expressed as the sum of the capital cost of the cooling tower (CC_{CT}), the capital cost of the steam condenser (CC_{SC}), and the capital cost of the cooling water pumps (CC_{CWPs}). The capital cost of the cooling water system includes cost for designing, purchasing materials, manufacturing components in factories, and cost for installing and building on the facility of the TPP.

$$CC_{\mathrm{CWS}} = CC_{\mathrm{CT}} + CC_{\mathrm{SC}} + CC_{\mathrm{CWPs}} \tag{5}$$

The annual investment cost of the cooling water system (AIC_{CWS}) is the cost related to the repayment of the loan taken for the construction of the cooling water system and is calculated according to the following formula [17]:

$$AIC_{\mathrm{CWS}} = CC_{\mathrm{CWS}} \cdot CRF \tag{6}$$

$$CRF = \frac{r \cdot (1+r)^n}{(1+r)^n - 1} \tag{7}$$

where, CRF is capital recovery factor; r is interest rate that is paid for loan repayment; n is number of years of loan repayment.

Formula (7) implies that the repayment (amortization) of the loan is made in equal annual installments (annuities).

The AOC_{CWS} is the cost of electricity for the operation of the CWPs corrected for savings or costs that arise with the change in the power of the LPST due to the change in the steam condensation pressure.

$$AOC_{\mathrm{CWS}} = \left(P_{\mathrm{CWPs}} - \Delta P_{\mathrm{LPST}}\right) \cdot \mathrm{IPUF} \cdot \tau \cdot \mathrm{LCOE} \tag{8}$$

where, P_{CWPs} is the power consumed by the CWPs, MW; ΔP_{LPST} is change in the LPST power with change in the steam condensation pressure, MW; IPUF is the installed power utilization factor; τ is the number of annual operating hours of the power plant, hr; LCOE is the levelized cost of energy produced by the power plant, € per MWh.

Inserting expressions (6), (7) and (8) into expression (4) yields the final expression for the objective function:

$$AC_{\mathrm{CWS}} = CC_{\mathrm{CWS}} \cdot \frac{r \cdot (1+r)^n}{(1+r)^n - 1} + \left(P_{\mathrm{CWPs}} - \Delta P_{\mathrm{LPST}}\right) \cdot \mathrm{IPUF} \cdot \tau \cdot \mathrm{LCOE} \tag{9}$$

5. Methodology

5.1. Mathematical Model

The mathematical model of a TPP cold end system consists of mathematical models for the following components: the cooling tower, the steam condenser, the low-pressure part of the steam turbine, and the circulating water pumps and pipelines. The goal of the mathematical model is to find a mathematical relationship between the process parameters and the dimensional parameters of the system equipment and to establish the mutual dependence of the decision variables based on the general principles of the laws of physics and generally accepted engineering calculation methods.

The mathematical model of a natural draft cooling tower consists of the mathematical model for the thermal calculation of the cooling tower and the mathematical model for aerodynamic calculation of the cooling tower. Water cooling in a wet cooling tower is the result of two physical processes: heat transfer by convection and mass transfer by evaporation. The intensity of these processes is determined by two laws of physics (Newton's law and Dalton's law) which form the basis for obtaining Merkel's differential equations on which the thermal calculation of the cooling tower is based [18].

$$\mathrm{d}Q = G_{\mathrm{cw}} \cdot c_{\mathrm{cw}} \cdot \mathrm{d}T_{\mathrm{cw}} = \beta_{\mathrm{xv}} \cdot \left(i'' - i\right) \cdot \mathrm{d}V \tag{10}$$

$$\mathrm{d}i = \frac{c_{\mathrm{cw}}}{\lambda} \cdot \mathrm{d}T_{\mathrm{cw}}\text{, where } \lambda = \frac{G_{\mathrm{a}}}{G_{\mathrm{cw}}} \tag{11}$$

In his approach, Merkel neglected the amount of water that evaporates in the cooling process. L. D. Bermam introduced a correction factor that considers the amount of water that evaporates, which is calculated by the formulas [19]:

$$k = 1 - c_{\mathrm{cw}} \cdot \frac{T_{\mathrm{cw2}}}{r_{\mathrm{Tcw2}}} \tag{12}$$

$$r_{\mathrm{Tcw2}} = r_0 - \left(c_{\mathrm{cw}} - c_{\mathrm{swv}}\right) \cdot T_{\mathrm{cw2}} \tag{13}$$

Considering equation (12), equation (11) can be written in the form:

$$\mathrm{d}i = \frac{c_{\mathrm{cw}}}{k \cdot \lambda} \cdot \mathrm{d}T_{\mathrm{cw}} \tag{14}$$

The solutions for equations (10) and (14), for the case of counterflow, as shown in **Figure 4**, can be written in the form:

$$Me = \int_{T_{\mathrm{cw2}}}^{T_{\mathrm{cw1}}} \frac{c_{\mathrm{cw}}}{i'' - i} \cdot \mathrm{d}T_{\mathrm{cw}} = \int_0^V \frac{\beta_{\mathrm{xv}}}{G_{\mathrm{cw}}} \cdot \mathrm{d}V \tag{15}$$

$$Me = \frac{c_{\mathrm{cw}} \cdot \left(T_{\mathrm{cw1}} - T_{\mathrm{cw2}}\right)}{6} \cdot \left[\frac{1}{i_1'' - i_2} + \frac{4}{i_{\mathrm{m}}'' - i_{\mathrm{m}}} + \frac{1}{i_2'' - i_1}\right] = A \cdot \lambda^n \cdot H_{\mathrm{CTf}} \tag{16}$$

$$i_2 - i_1 = \frac{c_{\mathrm{cw}} \cdot \left(T_{\mathrm{cw1}} \cdot T_{\mathrm{cw2}}\right)}{k \cdot \lambda} \tag{17}$$

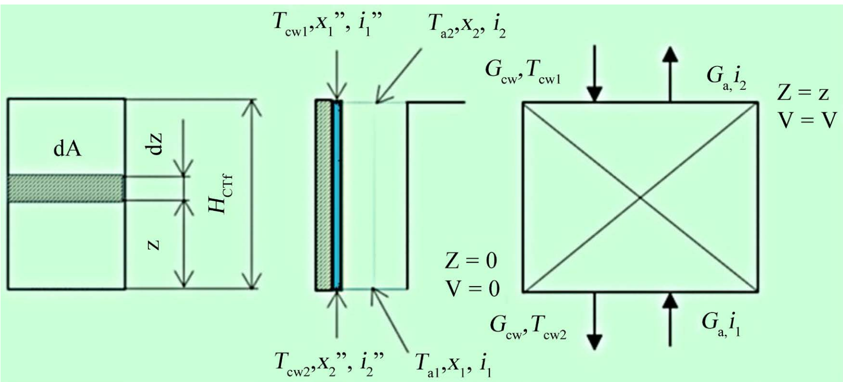

Figure 4. Change in water and air parameters in the wet cooling tower fill.

Expression (16) is obtained based on the following conditions and assumptions:

- numerical integration of $Me = \int_{T_{\mathrm{cw2}}}^{T_{\mathrm{cw1}}} \frac{c_{\mathrm{cw}}}{i'' - i} \cdot \mathrm{d}T_{\mathrm{cw}}$ using Simpson's rule [20]

where i_{m}" is enthalpy of saturated air at temperature $T_{\mathrm{cwm}} = \frac{T_{\mathrm{cw1}} + T_{\mathrm{cw2}}}{2}$, and

$$i_{\mathrm{m}} = \frac{i_1 + i_2}{2}$$

- βxv does not depend on the thermodynamic parameters of water and air, which was confirmed by tests,

- mass transfer coefficient in the cooling tower fill has the form:

$$\beta_{\mathrm{xv}} = A \cdot \left(\frac{G_{\mathrm{a}}}{A_{\mathrm{CTf}}} \right)^n \cdot \left(\frac{G_{\mathrm{cw}}}{A_{\mathrm{CTf}}} \right)^m \tag{18}$$

and considering that for most types of the cooling tower fills, m = 1 - n.

The aerodynamic calculation of a natural draft cooling tower is based on the condition that available draft and total air flow resistance in the cooling tower must be equal [21] [22].

$$H_{\mathrm{CTb}} \cdot \left(\rho_{\mathrm{a1}} \cdot \rho_{\mathrm{a2}} \right) \cdot g = \sum\nolimits_1^n \zeta_n \cdot \frac{v_n^2}{2} \cdot \rho_n \tag{19}$$

where, H_{CTb} is effective tower height of buoyancy, m; ρ_{a1} is air density at the CT entrance, kg/m^3; ρ_{a2} is air density at the CT exit, kg/m^3; g is acceleration due to gravity, m/s^2; ζ_{n} is local pressure loss coefficient; v_{n} is local air velocity, m/s; ρ_{n} is local air density, kg/m^3.

Since dominant airflow resistance is in the cooling tower fill, it is common to express the air flow resistance in the tower as a function of the air velocity in the tower fill (v_{CTf}) and the average air density in the tower fill (ρ_{am}) [21] [22]:

$$\sum\nolimits_1^n \zeta_n \cdot \frac{v_n^2}{2} \cdot \rho_n = \zeta_{\mathrm{t}} \cdot \frac{v_{\mathrm{CTf}}^2}{2} \cdot \rho_{\mathrm{am}} \tag{20}$$

$$\rho_{\mathrm{am}} = \frac{\rho_{\mathrm{a1}} + \rho_{\mathrm{a2}}}{2} \tag{21}$$

where, ζ_{t} is the total air flow resistance coefficient reduced to the air velocity in the tower fill.

If the air speed in the tower fill is expressed through the mass flow of air G_{a}, in kg/h, it can be written:

$$v_{\mathrm{CTf}} = \frac{G_{\mathrm{a}}}{3600 \cdot A_{\mathrm{CTf}} \cdot \rho_{\mathrm{am}}} \tag{22}$$

$$v_{\mathrm{CTf}} = \frac{\lambda \cdot G_{\mathrm{cw}}}{3600 \cdot A_{\mathrm{CTf}} \cdot \rho_{\mathrm{am}}} \tag{23}$$

If the water flow G_{cw}, in kg/h, is expressed through the parameter that represents the hydraulic water load of the tower fill q_{CTf}, in m^3/m^{2}h, the equation (23) is transformed into the following form:

$$v_{\mathrm{CTf}} = \frac{\lambda \cdot q_{\mathrm{CTf}} \cdot A_{\mathrm{CTf}} \cdot \rho_{\mathrm{cw}}}{3600 \cdot A_{\mathrm{CTf}} \cdot \rho_{\mathrm{am}}} \tag{24}$$

$$v_{\mathrm{CTf}} = \frac{\lambda \cdot q_{\mathrm{CTf}}}{3.6 \cdot \rho_{\mathrm{am}}} \tag{25}$$

From equations (19), (20), (21) and (25), the final expression for the effective tower height of buoyancy is obtained in the form:

$$H_{\mathrm{CTb}} = \zeta_{\mathrm{t}} \cdot \left(\frac{\lambda \cdot q_{\mathrm{CTf}}}{11.276}\right)^2 \cdot \frac{1}{\rho_{\mathrm{a1}}^2 - \rho_{\mathrm{a2}}^2} \tag{26}$$

The required total tower height can now be calculated [21]:

$$H_{\mathrm{CT}} = H_{\mathrm{CTb}} + 0.5 \cdot \left(H_{\mathrm{CTf}} + 0.5\right) + 0.75 \cdot H_{\mathrm{CTi}} \tag{27}$$

Expression (27) considers that the heat transfer in the cooling tower, in addition to the cooling fill, also occurs in the rain and spray zones, as pointed out by D. Bohn & K. Kusterer in chapter 6 of reference [23], *i.e.*, these zones are a part of the cooling fill.

The total air flow resistance coefficient ζ_t is calculated according to the methodology given in references [21] [22] [24] [25], where the local flow resistance coefficients in the cooling fill and drift eliminators are calculated according to the expressions given in references [26] and [27].

The mathematical model of the steam condenser is based on the heat transfer equation:

$$Q_{\mathrm{SC}} = U_{\mathrm{SC}} \cdot A_{\mathrm{SC}} \cdot \Delta T_{\mathrm{LMTD}} \tag{28}$$

$$\Delta T_{\mathrm{LMTD}} = \frac{\Delta T_{\mathrm{cw}}}{\ln \dfrac{\Delta T_{\mathrm{cw}} + \Delta T_{\mathrm{TTD}}}{\Delta T_{\mathrm{TTD}}}} \tag{29}$$

From equation (28), the area of the steam condenser (A_{SC}) is calculated. The number of condenser tubes (N_{SCt}) and the length of the condenser tubes (L_{SCt}) are calculated according to expressions (30) and (31), respectively [28]:

$$N_{\mathrm{SCt}} = 10^6 \cdot \frac{4}{\pi} \cdot \frac{G_{\mathrm{cw}} \cdot z}{\rho_{\mathrm{cw}} \cdot v_{\mathrm{SCt}} \cdot d_{\mathrm{ID}}^2} \tag{30}$$

$$L_{\mathrm{SCt}} = 10^3 \cdot \frac{A_{\mathrm{SC}}}{N_{\mathrm{SCt}} \cdot \pi \cdot d_{\mathrm{OD}}} \tag{31}$$

The average heat transfer coefficient of a steam condenser (U_{SC}) can be calculated according to the methodology given in references [28] and [29].

The purpose of the mathematical model of the LPST is to give the dependence of the turbine power change on the pressure in the SC, on the steam flow through the last stage of the turbine and on the design parameters of the last stage. In the most general case, this dependence can be obtained by a detailed calculation of the turbine, in various modes of operation. This is not usually done in optimization calculations. Either curves provided by the turbine manufacturer are used or a theoretical model is used that gives very good results according to measurements on objects. In this paper, the latter procedure was adopted, the base of which can be found in references [30]-[32]. Three cases of the turbine operating mode are characteristic: the subcritical flow region when $\Delta P_{\mathrm{LPST}} < 0$ (Equation 32), the supercritical flow region when $\Delta P_{\mathrm{LPST}} > 0$ (Equation 33), and the transitional flow region when $\Delta P_{\mathrm{LPST}} = 0$ [32].

$$\Delta P_{\mathrm{LPST}} = G_{\mathrm{s}} \cdot a^{*2} \cdot \left[\frac{1}{k-1} \cdot \left(1 - \varepsilon_k^{\frac{k-1}{k}}\right) \cdot \eta_{\mathrm{oi*}} - \frac{1}{2} \cdot \left(\varepsilon_k^{\frac{-2}{k}} - 1\right) + \frac{u \cdot \cos\beta_2}{a^*} \cdot \left(\varepsilon_k^{\frac{-1}{k}} - 1\right)\right] \tag{32}$$

$$\Delta P_{\text{LPST}} = G_{\text{s}} \cdot u \cdot a^{*} \cdot x \cdot \left\{ \left[\frac{k+1}{k-1} \left(1 - \frac{2}{k+1} \varepsilon_{k}^{\frac{k-1}{k}} \right) - \varepsilon_{k}^{\frac{-2}{k}} \cdot \sin^{2} \beta_{2} \right]^{\frac{1}{2}} - \cos \beta_{2} \right\} \tag{33}$$

$$A_{2} = D_{\text{m}} \cdot \pi \cdot l_{\text{STb}} \cdot \sin \beta_{2} \tag{34}$$

$$p^{*} = \frac{a^{*} \cdot G_{\text{s}}}{\mu_{2} \cdot k \cdot A_{2} \cdot N_{\text{LPST-ES}}} \tag{35}$$

$$\varepsilon_{\text{k}} = \frac{p_{\text{cond}}}{p^{*}} \tag{36}$$

$$u = \frac{D_{\text{m}} \cdot \pi \cdot n_{\text{r}}}{60} \tag{37}$$

where, G_s is steam flow rate, kg/s; a* is critical speed of sound, m/s: k is isentropic coefficient of steam in the LPST last stage; η_{oi^*} is internal efficiency coefficient of the LPST last stage; $\beta 2$ is exit steam velocity angle of the LPST last stage, ˚; x is moisture content of the steam exiting the LPST last stage, %; D_m is mean diameter of the LPST last stage, m; l_{STb} is blades length of the LPST last stage, m; μ_2 is flow coefficient of the LPST last stage; $N_{LPST\text{-}ES}$ is number of exit sections of the LPST; n_r is number of revolutions of the ST, rpm.

When steam expansion in a turbine reaches the "limit vacuum" (p_{cond} < p*), further lowering the condenser pressure doesn't increase power output from the LPST. This limit is determined by the following relationship [32]:

$$\varepsilon_{\text{kl}} = \sin \beta_{2}^{\frac{2k}{k+1}} \tag{38}$$

$$p_{\text{cond-}\Delta\text{PLPSTmax}} = p^{*} \cdot \varepsilon_{\text{kl}} \tag{39}$$

The main purpose of the mathematical model of cooling water pumps and cooling water pipelines is to calculate the power used to drive the cooling water pumps:

$$P_{\text{CWP}} = \frac{\rho_{\text{cw}} \cdot g \cdot Q_{\text{CWP}} \cdot H_{\text{CWP}}}{\eta_{\text{CWP}}} \tag{40}$$

$$H_{\text{CWP}} = H_{\text{CWPs}} + H_{\text{CWPd}} \tag{41}$$

The static head of the CWP (H_{CWPs}) measures the total vertical distance that the cooling water pump raises water in the cooling tower. The dynamic (frictional) head of the CWP (H_{CWPd}) is spent on overcoming frictional flow resistance in the system and can be expressed as the sum of the pump head spent on overcoming the flow resistance in the cooling water pipeline (ΔH_{CWPL}) and the flow resistance in the steam condenser (ΔH_{SC}). For the calculation of ΔH_{CWPL}, it is most convenient to use the Hazen-Williams empirical formula, which is intended for turbulent water flow. The ΔH_{SC} can be calculated according to the methodology given in references [28] and [29].

5.2. Computer Program

In the past, the main focus was to make the optimization calculation procedures for an engineering system as efficient as possible to save computer time, but now this approach has little meaning, and more importance is given to simplicity of implementation. Based on this premise, the exhaustive search method [33] [34] was used in this study for optimal design of the recirculating cooling water system of TPPs. The main strength of the exhaustive search method is that it is guaranteed to find the optimal solution from among the domain specified for the decision variables.

For the numerical solution of the system of nonlinear equations defined in the mathematical model and the objective function, the computer program (written in FORTRAN) has been developed. It consists of the following components: a main program, a subroutine for the cooling tower thermal calculation, a subroutine for the cooling tower aerodynamic calculation, a subroutine for calculating the temperature of the cold water in the cooling tower as a function of the plant location and prevailing atmospheric conditions, a subroutine for the steam condenser sizing, a subroutine for calculating the steam condensation pressure, a subroutine for calculating the power of the low-pressure part of the steam turbine, and several subroutines for calculating thermodynamic properties of fluids. **Figure A1** in Annex shows the algorithm of the main computer program based on the exhaustive search algorithm for solving the optimization problem.

In the program, the decision variables (T_{app}, ΔT_{cw}, q_{CTf}, H_{CTi}, H_{CTf}, ΔT_{TTD}, and v_{SCt}) vary between the lower and upper bounds with a variation step fixed for each variable. The AC_{CWS} is calculated for each variation of the decision variables in an iterative procedure. The AC_{CWS} calculated in the previous iteration (AC_{CWSpi}) is compared with the AC_{CWS} in the next iteration (AC_{CWSni}), and only the smaller value is memorized so that at the end of the iterative procedure the minimum value of the annual costs of the cooling water system AC_{CWSmin} is obtained, which corresponds to the optimal values of the decision variables ($\Delta T_{app\text{-}opt}$, $\Delta T_{cw\text{-}opt}$, $q_{CTf\text{-}opt}$, $H_{CTi\text{-}opt}$, $H_{CTf\text{-}opt}$, $\Delta T_{TTD\text{-}opt}$, and $v_{SCt\text{-}opt}$).

The exhaustive search algorithm has been streamlined (by adjusting the variation step for all the decision variables) to prune the discretized search space with the intention to remediate the time and space complexity of the algorithm.

All the decision variables, at the end of the optimization process, should be between the lower and upper set values, apart from the parameters of ΔT_{app} and ΔT_{TTD}, whose minimum values are included in the optimization constraints. If one or more of the decision variables, which are not included in the optimization constraints (ΔT_{cw}, q_{CTf}, H_{CTi}, H_{CTf}, and v_{SCt}), at the end of one iteration cycle are at the lower or upper bounds of their given values, then their given bounds are moved until all the decision variables fall within their given intervals. To achieve this, it usually takes several cycles because it often happens that when one of the decision variables "falls" within the limits of the given interval, it "drives" one or more of the other decision variables to their lower or upper limit. For this reason, at the end, the variation step for all the decision variables was reduced to 0.1.

5.3. Validation Process

The existing 300 MW TPPs Gacko and Ugljevik with the steam turbine type K-

300-240 LMZ and the cold end system components (CT, SC, LPST, and CWPs) of known design parameters and dimensional characteristics were used in the validation process of the proposed optimization model.

The mathematical model and computer program were checked in such a way that the calculation results of the cold end system components were compared with their actual characteristics for the same design and operating conditions. The results are found to be conforming and accurate. In addition, based on the author's extensive design experience, the optimal values of the decision variables and optimal sizes of the cold end system equipment are within the expected ranges.

6. Capital Cost Functions

The capital cost functions of the TPP cooling system equipment used for the purpose of optimization must be directly or indirectly related to the operating and dimensional parameters of the equipment. The purchased equipment's capital cost must include design and project management costs, production costs, transportation costs, and assembly/installation/construction costs at the facility. The more accurate the equipment cost functions are, the more accurate and reliable the optimization results are for use.

The best way to find the equipment capital cost (investment costs) is if the purchased equipment cost is evaluated based on vendor quotations, the data bank of the designing company, or experience from previous projects. On the other hand, some empirical correlations have been developed to estimate approximate values of purchased equipment costs when there are no sources of data. This kind of empirical cost equation is useful for modeling and optimization tasks, as these equations are user-friendly for such tasks.

The empirical correlations for the capital cost estimate of the cooling water system components, presented in the literature, are shown in **Table 1**.

Table 1. Capital cost functions.

Component	Capital cost estimate formulas (€)	Ref.
	Natural draft cooling tower shell	
	$CC_{CTshell} = (0.98 - 0.595 \cdot 10^{-2} \cdot H_{CT} + 0.6 \cdot 10^{-4} \cdot H_{CT}^2 - 0.0217 \cdot D_{CTim} + 0.76 \cdot 10^{-3} \cdot H_{CT} \cdot D_{CTim}) \cdot 10^6 \cdot CCF_{CTshell}$, $CCF_{CTshell} = 2.91$	[15] [37]
	Natural draft cooling tower fill	
	$CC_{CTfill} = C1 \cdot V_{CTfill} \cdot CCF_{CTfill}$, $C1 = 250\ \$/m^3$, $CCF_{CTfill} = 1.0$	[38]
	Steam condenser	
	$CC_{SC} = [C61 \cdot A_{SC} \cdot (2200/U_{SC}) + C62 \cdot G_{cw}] \cdot CCF_{SC}$, $C61 = 280.74\ \$/m^2$, $C62 = 746\ \$/(kg/s)$, $CCF_{SC} = 1.05$	[39]
	Cooling water pump	
	$CC_{CWP} = C71 \cdot P_{CWP}^{0.71} \cdot [1 + 0.2/(1 - \eta_{CWP})] \cdot CCF_{CWP}$, $C71 = 705.48\ \$/kW$, $CCF_{CWP} = 2.85$	[39]

The cost correction factors (CCF) for each type of equipment are determined considering the following conditions: the conversion of prices expressed in US$ into prices in €, the inflation price index, the price ratio of individual components of the cooling water system to be within realistic limits [35], the price ratio of the cooling water system in relation to the total price of the convectional power plant to be within realistic limits, and the experience of previous projects [35] [36].

7. Case Studies

This article is organized into several parts to illustrate the application of the proposed optimization method using case studies. The case studies are related to the cold end system of a 300 MW TPP. The objective of the studies is to find an optimal design of the system that will perform its task at the lowest possible annual cost (capital and operating) while satisfying the specified input design and operating conditions as well as the imposed constraints.

7.1. Case Study 1

In this part (Part I) of the article, Case Study 1 is presented as the base case study.

7.2. Input Design/Operating Conditions and Constraints for Case Study 1

Input design and operating conditions for the SC are:
- Nominal installed power: P_{ST} = 300 MW
- Steam flow to the SC: G_s = 173.913 kg/s
- Installed power utilization factor: IPUF = 0.85
- Number of exit sections of the LPST: $N_{LPST\text{-}ES}$ = 3
- Mean diameter of the LPST last stage: D_m = 2.480 m
- Blades length of the LPST last stage: l_{STb}= 0.960 m
- Number of revolutions of the ST: n_r = 3000 rpm
- Exit steam velocity angle of the LPST last stage: β_2 = 35˚
- Isentropic coefficient of steam in the LPST last stage: k = 1.135
- Critical speed of sound of the LPST last stage: a* = 370 m/s
- Flow coefficient of the LPST last stage: μ_2 =0.98
- Internal efficiency coefficient of the LPST last stage: η_{oi*} = 0.87
- Moisture content of steam exiting the LPST last stage: x = 0.92

● Input design and operating conditions for the SC are:
- Heat load: Q_{SC} = 400 MW
- Tubes diameter: d_{OD}/d_{ID} = 28/26 mm
- Number of cooling water paths: z = 2
- Heat transfer surface cleanliness factor: a = 0.8

● Input design and operating conditions for the CT are:
- Heat load: Q_{CT} = 400 MW
- Design cold water temperature: T_{cwc} = 29.0˚C
- Coefficients characterizing CT fill: A = 1.5, n = 0.5

- Input design and operating conditions for the CWPs and the CWPLs:
- Equivalent length of the CWPL & fittings: $L_{CWPL} = 750$ m
- Cooling water velocity in the CWPLs: $v_{CWPL} = 2.25$ m/s
- Number of the CWPLs: $N_{CWPLs} = 2$
- Number of the CWPs: $N_{CWPs} = 3$ (2 + 1)
- Efficiency of the CWP: $\eta_{CWP} = 0.85$
- Efficiency of the CWP motor: $\eta_{CWPm} = 0.95$
- Hazen-Williams friction loss coefficient for the CWPLs: C =110
- Static head of the CWPs: $H_{CWPs} = H_{CTi} + H_{CTf} + 2.5$ m
- Input data for ambient air conditions for the thermal power plant location:
- Barometric pressure: $P_b = 1.0$ bar
- Average annual ambient wet bulb temperature: $T_{wb\text{-}amb} = 5.6$˚C ($T_{db\text{-}amb} = 8$˚C @ RH = 70%)
- Input data for calculating the annual operating costs of the CWS:
- Interest rate for loan repayment: r = 8%
- Number of years of loan repayment: n = 30
- Levelized cost of electricity: LCOE = 100 € per MWh
- Constraints for the decision variables:
- $\Delta T_{app} \geq 5.0$ K
- $\Delta T_{TTD} \geq 3.0$ K
- Constraints for the cooling tower dimensional variables:
- $(D_{CTt}/D_{CTft})^2 = 0.375$ ($D_{CTt} = 0.613 \cdot D_{CTft}$)
- $H_{CTt\text{-}e} = 0.25 \cdot (H_{CT} - H_{CTi})$
- $H_{CTfb\text{-}t} = 0.75 \cdot (H_{CT} - H_{CTi})$
- $1.2 \cdot D_{CTb} < H_{CT} < 1.4 \cdot D_{CTb}$
- $A_{CTi}/A_{CTf} = (2 \cdot \pi \cdot R_{CTf} \cdot H_{CTi})/(R_{CTf}^2 \cdot \pi) = 2 \cdot H_{CTi}/R_{CTf} \geq 0.35$
- $\alpha = 72$˚

Notes:

1) The values of the above-listed input data and constraints are chosen to be as realistic as possible.

2) Constraints of the process and dimensional parameters of the system equipment resulting from the balance of mass and energy and the general laws of physics, as stated in the mathematical models, are implied and are not specifically stated here.

8. Numerical Results and Comments

Based on the input data given in section 7.2 above, the optimal results for the decision variables and equipment sizes of the cold end system components are presented in Tables 2-5. The five optimization cases show the impact of the cooling water approach to the ambient wet bulb temperature (ΔT_{app}) on the optimization results.

Sensitivity analysis is utilized to assess the sensitivity of the objective function (AC_{CWS}) with respect to the change in the decision variables. The results are shown in **Figure 5**. The optimal case, where all decision variables have optimal values, was taken as the reference case. It can be seen from the figure that the hydraulic water load of the tower fill (q_{CTf}) has the greatest influence, and the water velocity in the condenser tubes (v_{SCt}) has the least influence on the AC_{CWS}.

Table 2. Optimal values of the decision variables.

ΔT_{app} (K)	ΔT_{cw} (K)	q_{CTf} (m^3/m^2h)	H_{Cti} (m)	H_{CTf} (m)	ΔT_{TTD} (K)	v_{cond} (m/s)	AC_{CWS} (€)
5.0	7.5	9.1	9.4	1.6	3.0	1.3	3,298,517.30
5.5	7.5	9.2	9.3	1.4	3.0	1.3	3,458,124.80
6.0	7.4	9.1	9.2	1.2	3.0	1.3	3,654,088.00
6.5	7.0	9.4	9.2	1.1	3.0	1.3	3,848,469.50
7.0	6.7	9.6	9.1	1.0	3.0	1.3	4,058,265.30

Table 3. Optimal values of the p_{cond}.

ΔT_{app} (K)	T_{cwc} (˚C)	ΔT_{cw} (K)	ΔT_{TTD} (K)	T_{cond} (˚C)	p_{cond} (kPa)	ΔP_{LPST} (MW)	P_{CWPs} (MW)
5.0	16.5	7.5	3.0	27.0	3.58	2.841	2.665
5.5	17.0	7.5	3.0	27.5	3.68	2.508	2.616
6.0	17.5	7.4	3.0	27.9	3.77	2.248	2.598
6.5	18.0	7.0	3.0	28.0	3.77	2.221	2.712
7.0	18.4	6.7	3.0	28.1	3.81	2.117	2.785

Table 4. Optimal dimensions of the CT.

ΔT_{app} (K)	H_{CT} (m)	H_{Cti} (m)	H_{CTf} (m)	H_{CTft-t} (m)	H_{CTt-e} (m)	D_{CTb} (m)	D_{CTft} (m)	D_{CTt} (m)	D_{CTe} (m)
5.0	104.8	9.4	1.6	70.0	23.9	87.4	80.1	49.1	53.6
5.5	104.5	9.3	1.4	70.0	23.8	86.7	79.7	48.8	53.3
6.0	105.3	9.2	1.2	70.9	24.0	87.5	80.6	49.4	53.9
6.5	106.0	9.2	1.1	71.5	24.2	88.3	81.6	50.0	54.6
7.0	107.1	9.1	1.0	72.5	24.5	89.1	82.5	50.6	55.2

Table 5. Optimal parameters of the SC and CWPs.

ΔT_{app} (K)	A_{SC} (m^2)	N_{SCt}	L_{SCt} (m)	ΔH_{SC} (mH_2O)	z	ΔH_{CWPL} (mH_2O)	H_{CWP} (mH_2O)	Q_{CWP} (m^3/s)	P_{CWP} (MW)
5.0	27,711	36,950	8.5	2.1	2	1.6	17.2	6.4	1.333
5.5	27,512	36,954	8.5	2.1	2	1.6	16.9	6.4	1.308
6.0	27,504	37,456	8.3	2.1	2	1.6	16.6	6.5	1.299
6.5	28,156	39,600	8.1	2.0	2	1.5	16.4	6.8	1.356
7.0	28,623	41,376	7.9	2.0	2	1.5	16.1	7.1	1.392

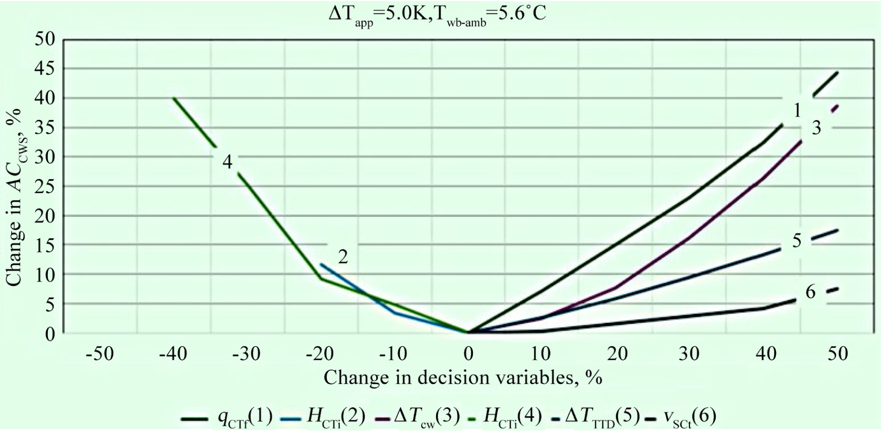

Figure 5. Sensitivity analysis at an electricity price of €100 per MWh.

9. Conclusions

The cold end system remains the only part of TPPs whose parameters and dimensions are not subject to standardization. Climatic and economic specificities of the location of the plant are the basic elements that should determine optimal characteristics of the system in each project.

There are several contributions this study makes:

a) It includes the optimization of the seven most influential parameters that characterize the design and operation of the TPP cold end system.

b) It can be used for different types of TPPs (conventional coal-fired TPPs, NPPs, and various types of power plants for combined production of electricity). The specificities of the TPP type are determined through the input parameter for LCOE, which also reflects the specificities of the environmental protection standards for each TPP type.

c) It can be used for designing new cooling water systems and for optimizing the operation of existing cooling water systems.

d) Global optimization of the entire cooling water system can easily be reduced to partial optimization of the system.

e) The proposed optimization method has a global character in such a way that the input parameters consider the climatic and economic specificities of the geographical location of the thermal power plant as well as the specificities of various local and global environmental protection standards.

f) The developed optimization model can be modified to consider the specifics of the mathematical and economic models of the companies involved in the design and construction of the cold end system components so that the optimal solution of general interest is reached on a specific project, including equipment suppliers, investors, and the end user of the thermal power plant.

g) By applying the proposed optimization model, it is possible to make significant (measured in millions of €) of savings in the operation of a TPP on an annual basis. The greater the installed power of the TPP, the greater the savings.

The optimization methodology presented in this paper opens several avenues for future research: similar models can be developed for other types of power plant cooling systems, such as systems with dry cooling towers, wet cooling towers with natural draft and cross flow, and wet mechanical draft cooling towers. A very interesting topic for further research would be the joint work of mechanical and civil engineers and researchers to arrive at the optimal design of a natural draft cooling tower, both from the aspect of thermodynamics and aerodynamics of the tower as well as from the aspect of civil design and construction methods.

Conflicts of Interest

The author declares no conflicts of interest regarding the publication of this paper.

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Nomenclature

Symbol	Definition	Unit
A	Area	m^2
c	Specific heat capacity	J/kg K
D, d	Diameter	m
D_m	Mean diameter of LPST exit stage	m
D_{CTb}	Diameter of CT at base	m
D_{CTe}	Diameter of CT at exit	m
D_{CTfb}	Diameter of CT at fill base	m
D_{CTft}	Diameter of CT at fill top	m
D_{CTim}	Diameter of CT at middle of air inlet height	m
D_{CTt}	Diameter of CT at throat	m
g	Acceleration due to gravity	m/s^2
G	Flow rate	kg/s
H	Height or CWP head	m or mH_2O
H_{CT}	Height of CT, m	m
H_{CTf}	Height of CT fill	m
H_{CTi}	Height of CT air inlet	m
$H_{CTft\text{-}t}$	Height of CT from top of fill to throat	m
$H_{CTfb\text{-}t}$	Height of CT from bottom of fill to throat	m
$H_{CTt\text{-}e}$	Height of CT from throat to exit	m
i	Enthalpy	J/kg K
L	Length	m
l_{STb}	Length of blades of LPST exit stage	m
Me	Merkel number	-
m	CT fill parameter	-
N, n	Number of years or number of units, or CT fill parameter	-
p	Pressure	N/m^2
P	Electric power	MW
q	Hydraulic water load on CT fill	m^3/m^2h
Q	Heat transfer rate or flow capacity of CWPs	W or m^3/s
T	Temperature	°C, K
r	Interest rate on capital	%
r_T	Heat of vaporization of water at temperature T	J/kg
s	Entropy	J/kg K
U	Heat transfer rate	W/m^2K
v	Velocity	m/s
V	Volume	m^3

Continued

x	Moisture content of steam at LPST exit	%
z	Number of cooling water paths in SC	-
	Abbreviations	
AC	Annual cost	
AIC	Annual investment cost	
AOC	Annual operating cost	
CC	Capital cost	
CRF	Capital recovery factor	
CT	Cooling tower	
CWP	Cooling water pump	
CWPL	Cooling water pipeline	
CWS	Cooling water system	
ES	Exit section	
ID	Inside diameter	
IPUF	Installed power utilization factor	
ITD	Initial temperature difference	
LCOE	Levelized cost of energy	
LMTD	Logarithmic mean temperature difference	
LPST	Low pressure steam turbine	
NPP	Nuclear power plant	
OD	Outside diameter	
RH	Relative humidity	
SC	Steam condenser	
ST	Steam turbine	
TPP	Thermal power plant	
TTD	Terminal temperature difference	
	Subscripts	
a	Air	
amb	Ambient	
app	Approach	
cond	Condensation	
cw	Cooling water	
cwc	Cooling water cold	
db	Dry bulb	
ni	Next iteration	
opt	Optimal status	
pi	Previous iteration	

Continued

s	Steam	
swv	Superheated water vapor	
wb	Wet bulb	
“	Condition at phase (liquid/gas) interface	
	Greek symbols	
α	Angle of rise of lower part of tower shell	°
β_2	LPST exit steam velocity angle	°
β_{xv}	CT volumetric mass transfer coefficient	kg/m^{3}h
μ_2	LPST steam flow coefficient	-
η	Efficiency	%
ρ	Density	kg/m^3
λ	ratio of air mass flow to cooling water mass flow in CT	-
τ	Annual operating hours	hr
ζ	Pressure loss coefficient	-
Δ	Increment	-

Annex

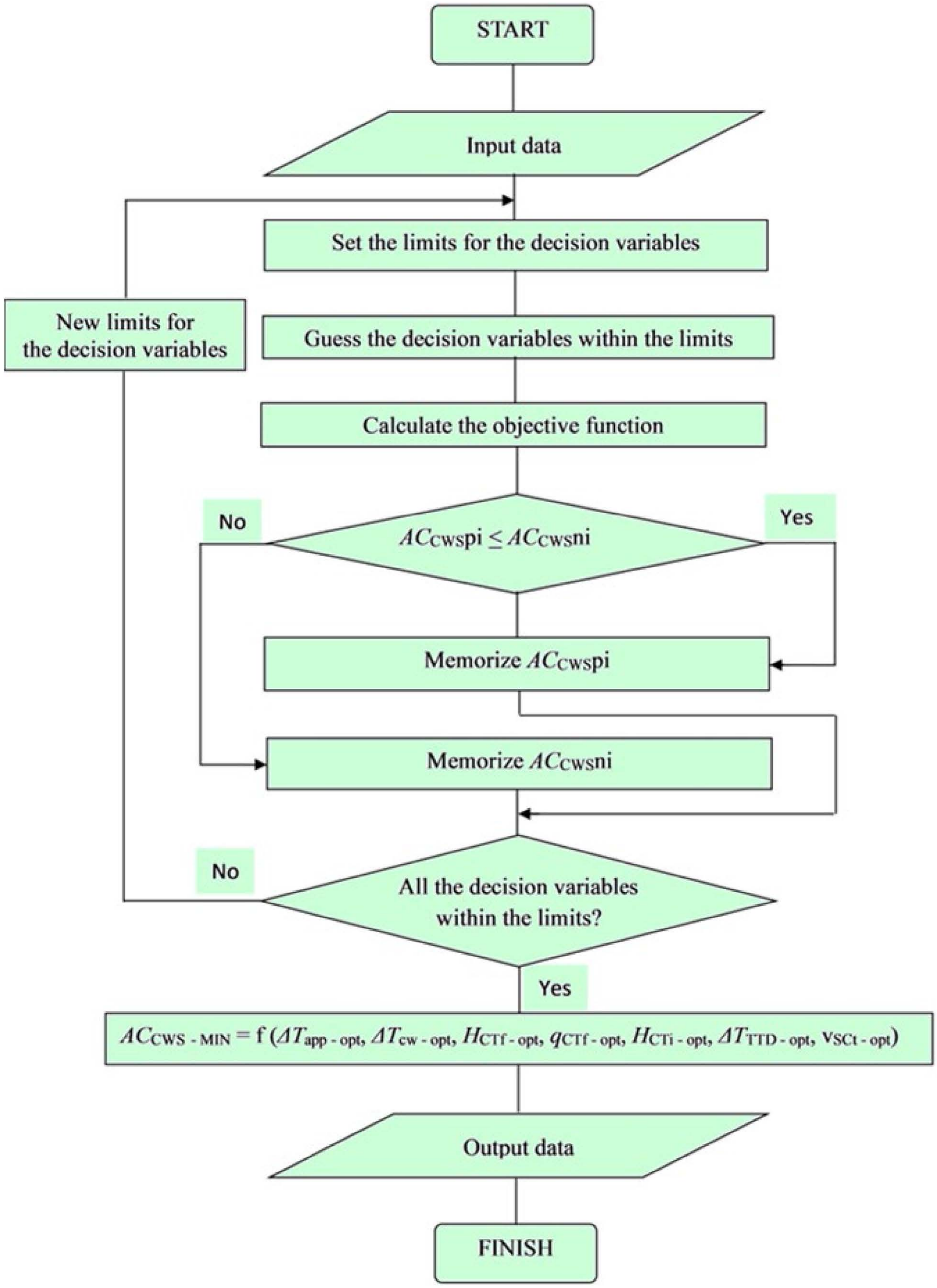

Figure A1. The algorithm of the optimization process.